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Fatigue Life Assessment for Metallic Structure: A Case Study of Shell Structure under Variable Amplitude Loading

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Abstract: This study presents an analysis technique to asses the fatigue life of a shell structure under variable amplitude loadings (VAL). For this purpose, the finite element analysis technique was used for the simulation works. The life prediction results are useful for improving the component design methodology at the early developing stage especially for shell structures such as a pressure vessels or pipelines analysis. The fatigue life prediction was performed using the finite element based fatigue analysis codes. In addition the ability of stress-life (S-N) and strain-life (ε-N) approaches to correlate and predict life are examined according to the different damage and failure rules. Numerical life prediction results (S-N and ε-N) of shells under VAL, as well as Constant Amplitude Loading (CAL) are presented and discussed. The effect of the mean stress, surface finish and the shell thickness are studied and discussed as apart of the interactions between geometries, loadings and materials. The simulation results showed that more studies on the shell structure need to be performed in order to obtain more accurate fatigue life.

Key words: Fatigue, finite element analysis, life prediction, shell, variable amplitude loading

INTRODUCTION

Many shell structures such as pressure vessels or pipelines, contain defects from the manufacturing and servicing processes. These defects can affect the safety of the structure and even depress their service life; may lead to enormous economic costs and jeopardize the surrounding ecological environments. There is a continuing tendency for damage-induced accidents in the shell structure; consequently, appropriate assessment methods of the influence of damage on the safety are needed. An important point in fracture assessment of shells is that the stresses in them are mainly of membrane type (tension or compression), even for the case subjected to global bending moments. Traditionally, three-dimensional solid finite elements are employed in discretising the shell structure in order to account for the crack.

Fatigue life prediction is still very much an empirical art rather than a science, despite being a relatively old subject having nearly 150 years of history, described in a number of books or review papers (Schutz, 1996). In the old days, the application of applied stresses (S) vs. fatigue life cycles (N) curves were measured and their empirical nature generally maintained even when simplified equations like the Basquin (1910) power law emerged, or when recognizing simpler rules for the fatigue limit, or various other effects on the basic fatigue S-N curve, like the mean stress, roughness, notch, dimension. This was generally driven by the need to have engineering rules still used when designing against fatigue with the safe-life approach, i.e., virtually for infinite life.

In a sense, most of the present design processes still use the empirical approaches of the pre-Paris era, particularly for long-lives (High Cycle Fatigue (HCF)), for which a damage tolerant approach is very difficult to implement, since cracks grow very slowly and are not detected for a large part of their life. It is still difficult today, even with the most advanced models, material data information and computational capabilities, to derive an entire S-N curve from integration of crack propagation laws, particularly because of various regimes of deviations from the standard Paris law regime, in particular due to short cracks many structural components are subjected to cyclic loading. In these components, fatigue damage is the prime factor in affecting structural integrity and service life. Fatigue damage is typically divided into three stages: crack initiation, crack propagation and final failure. These three stages are important in determining the fatigue life of structural components (Pugno *et al*., 2006).

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Many shell structures are subjected to complex cyclic loading spectrums ranging from small vibrations to large load excursions. To conduct structural reliability analyses of the structures, fatigue crack initiation and growth properties are crucial material data input parameters. Moreover, an understanding of the fatigue crack initiation and growth characteristics of the required shell grade materials is essential to evaluate useful life.

A fatigue life analysis based on the stress-life and strain-life approaches were followed to obtain an estimate of the expected life of the shell structure in this study, with application of VAL. The theoretical studies of fatigue life prediction are to be established with application of different theories in both S-N and ε-N approaches, using finite element analysis for modeling and simulation based fatigue analysis codes with a rainflow counting method. Then the analysis of the modeling results of using CAL and VAL will be discussed in view of above scope. To show the interaction between geometry, loading and material; the effect of using CAL instead of VAL will be studied and factors affecting the fatigue life prediction will studied also such as, mean stress, shell thickness and surface roughness using different theories. More calculation needs to be done in the future in order to optimize these factors and to study the effect of other factors such as, corrosion, residual stresses, temperature change, heat treatment, etc.

MATERIALS AND METHODS

In Germany during the 1850s and 1860s, August Wöhler, performed many laboratory fatigue tests under repeated stresses. These experiments were concerned with railway axle failures using stress versus life (S-N) diagram as being reported in Schutz (1996) and Cui (2002). For a long time such curves were labeled as a Wöhler curve instead of the now more frequently used term S-N curve. In the 20th century numerous fatigue tests were carried out to produce large numbers of S-N curves. The fatigue design curves in the American Society of Mechanical Engineers (ASME) Code are based on simplified approaches developed more than 25 years ago. In particular, the Stress-Life (S-N) data initially included in Section XI of the ASME Boiler and Pressure Vessel Code (Liaw *et al*., 1997) relative to pressure retaining materials for vessels utilized in nuclear applications was obtained in the same time and still to be updated in order to improve the fatigue design methodology for more accurate life prediction analyses of shell structure components.

Langer (1937) first proposed to separate the fatigue damage process into two stages of crack initiation and crack propagation. In 1950's Coffin (1954) expressed the linear damage rule (LDR) in terms of plastic strain range, which is related to fatigue life through the Coffin-Manson relation. In a later study, Topper and Biggs (1966) used the strain-based to correlate their experimental results. A review on the applications of the LDR to strain-controlled fatigue damage analysis was given by Miller (1970).

However, research on the influence of defects on the fatigue life of shells is limited. Fowler *et al*. (1995) employed full-scale tests to evaluate the fatigue life of shells with defects subjected to cyclic internal pressure. Also the fatigue crack initiation criterion for the assessment of the residual life of shells (gas transmission pipelines) with defects was studied by Zheng *et al*. (2005). Early works on fatigue theories involved correlation of fatigue life to the maximum principal strain range, the maximum shear strain range and the maximum octahedral shear strain range. Brown and Miller (1982) proposed a multiaxial fatigue theory based on the physical interpretation of the mechanisms of fatigue crack growth. A rational and quantitative formula for assessing the effect of a defect on the fatigue life of shells with mechanical damage has not yet been proposed. There are at least 50 models for cumulative fatigue damage and life prediction (Zheng and Wang, 1997), none of which has been generally accepted. Therefore it is still necessary to study the fatigue accumulation processes further.

The fatigue-life of a component can be modeled using the level-crossing approach assuming that the crack growth can be modeled with Paris law (Josefson *et al*., 2000), also using effective strain-life and E $\Delta\varepsilon$-life data (Topper and Lam, 1997), Crack Tip Opening Displacement (CTOD) (Skallerud *et al*., 2006). The crack-opening stress level corresponding to the stress cycle which reduces crack-opening stress to a level reached by 1/200 of the number of cycles, commonly used approach of cutting the loads as cycles (Khalil and Topper, 2003). Some authors used commercial software to analyze local cyclic elastic-plastic stress-strain responses that will be use for estimating fatigue life (Li *et al*., 2006; Glancey and Stephens, 2006).

Though many damage models have been developed, unfortunately, none of them enjoys universal acceptance. Each damage model can only account for one or several phenomenological factors, such as load dependence, multiple damage stages, non-linear damage evaluation, load sequence and interaction effects, overload effects, small amplitude cycles below fatigue limit and mean stress factors. The applicability of each model varies from case to case and the Palmgren-Miner LDR (Fatemi and Yang, 1998) is still dominantly used in design, in spite of its major shortcomings. Also, the most common method for cumulative damage assessment using Linear Elastic Fracture Mechanics (LEFM) has been based on integration of a Paris-type crack growth rate equation, with modification to account for load ratio and interaction effects. More efforts in the study of cumulative fatigue damage are needed in order to provide design engineers with a general and reliable fatigue damage analysis and life prediction model.

Fatigue-life prediction under variable amplitude loading is normally based on the stress history and some assumption regarding the damage process. The established models for fatigue damage accumulation are based on the linear accumulation rule originally proposed by Palmgren (1924) and the damage caused by each load is determined by the experimental Wöhler curve (Fatemi and Yang, 1998; Dowling, 1988). In the previous study by Savaidis *et al.* (2002) cyclic loading components with constant amplitude have been assumed to simplify the analysis. An approximate model has been developed for a notched cylinder under fatigue loading with variable amplitude in many studies as a constant load to simplify the analysis. Engineering structures under service conditions, however, are often undergone variable amplitude fatigue loading, which make the problem much more complex but has to be considered in a realistic fatigue life assessment for multiaxial variable amplitude fatigue loadings have been proposed in recent years (Zhen and Baotong, 1996; Topper and Lam, 1997).

In fatigue testing, the test specimen is subjected to alternating loads until failure. The loads applied to the specimens were defined by either a constant stress range (S_r) or a constant stress amplitude (S_k). The stress range and stress amplitude are defined as Eq. 1 and 2, respectively.

$$S_r = S_{max} - S_{min} \tag{1}$$

$$S_k = \frac{S_r}{2} = \frac{(S_{max} - S_{min})}{2} \tag{2}$$

The magnitude of the stress range or amplitude is the independent variable and the number of cycles to failure is the dependent variable. The number of cycles to failure is the fatigue life (N_f), as the form of cycles are shown in Fig. 1.

Most of the time, S-N fatigue testing is conducted using fully reversed loading. Fully reversed indicates that loading is alternating about a zero mean stress. The mean stress, stress ratio and amplitude ratio are mathematically defined in Eq. 3, 4 and 5, respectively.

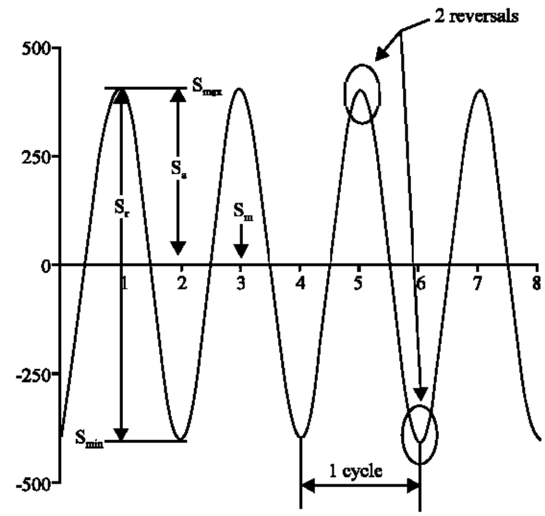

Fig. 1: Symbols used with the cyclic stresses and cycles

$$S_m = \frac{(S_{max} + S_{min})}{2} \tag{3}$$

$$R = \frac{S_{min}}{S_{max}} \tag{4}$$

$$A = \frac{S_a}{S_m} = \frac{1 - R}{1 + R} \tag{5}$$

Equation 6 represents the typical S-N curve:

$$S_k = S'_f (2N_f)^b \tag{6}$$

where, b is the fatigue strength exponent and S'_f is the fatigue strength coefficient. This expression developed from log-log S-N graphs is the most widely used equation (also known as the Basquin relation) in the stress-based approach to fatigue analysis and design (Dowling, 1988).

Fatigue life data exhibit widely scattered results due to the inherent micro-structural in homogeneity in the materials properties, differences in the surface and the test conditions of each specimen and other factors. In general, the variance of log life increases as the stress level decreases. It has been observed that once grains nucleate cracks in a material at high stress levels, these cracks have a better chance of overcoming the surrounding microstructure. Most of the grains can successfully nucleate cracks at low stress levels, but only very few of them can overcome the surrounding obstacles such as orientation, size and microstructure to grow a crack. As a result of the unavoidable variation in fatigue data, median S-N fatigue life curves are not sufficient for fatigue analysis and design. The statistical nature of the fatigue must be considered.

The fatigue damage of a component correlates strongly with the applied stress amplitude or applied stress range and is also influenced by the mean stress. In the high-cycle fatigue region, normal mean stresses have a significant effect on fatigue behaviour of components. Normal mean stresses are responsible for the opening and closing state of micro-cracks. Due to the opening of micro-cracks that accelerates the rate of crack propagation and the closing of micro-cracks retards the growth of cracks, tensile normal means stresses are detrimental and compressive normal mean stresses are beneficial in terms of fatigue strength. The shear mean stress does not influence the opening and closing state of micro-cracks and not surprisingly, has little effect on crack propagation. There is very little or no effect mean stress on fatigue strength in the low-cycle fatigue region in which the large amounts of plastic deformation erases any beneficial or detrimental effect of a mean stress. Early empirical models were proposed to compensate for the tensile normal mean stress effects on the high-cycle fatigue strength.

Gerber (1874) proposed a parabolic representation of Wöhler's fatigue limit data on a plot of as shown in Fig. 2. Goodman introduced a theoretical line representing the available fatigue data based on an impact criterion for bridge designs. Goodman justified the use of the impact criterion on a basis that it was easy, simple to use and provided a good fit to the data. Haigh (1917) first plotted fatigue data for brasses on a S_k vs. S_m plot. Figure 3, represents the Haigh plot of Gerber and Goodman mean stress corrections. The ordinate of the Haigh plot is the normalized fatigue limit and the maximum mean stress is limited to the ultimate strength S_u. The curve connecting these two points on the two axes represents combinations of stress amplitudes and means stresses given at the fatigue limit life.

Mathematically, the Gerber parabola and the Goodman line in Haigh's coordinates (Li *et al.*, 2006) can be expressed as the expressions in Eq. 7 and 8. It mentioned about the Gerber's mean stress correction, in a form of:

$$S_e = \frac{S_k}{1 - \left(\frac{S_m}{S_u}\right)^2} \tag{7}$$

and the Goodman's mean stress correction is mathematically defined as:

$$S_e = \frac{S_k}{1 - \frac{S_m}{S_u}} \tag{8}$$

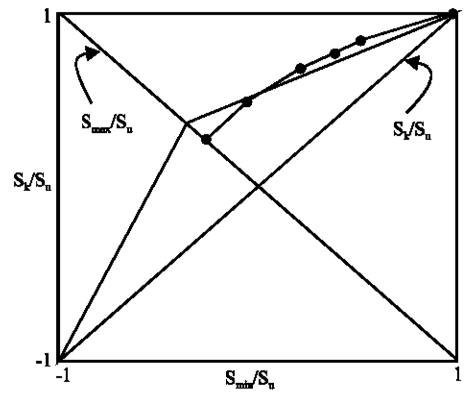

Fig. 2: Gerber's and Goodman diagrams for Wohler's data

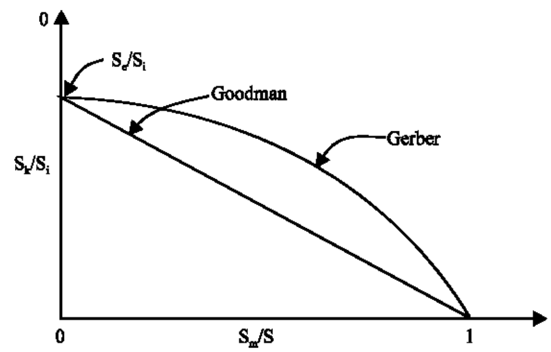

Fig. 3: Diagrammatic Haigh's plot for the Gerber's and Goodman's curves

where, S_e is the fatigue limit for fully reversed loading, S_k is the stress amplitude and S_m is the mean stress. The component design and analysis using the stress-based fatigue life prediction techniques were also presented in this section. This approach to the fatigue analysis of components works well for situations in which only elastic stresses and strains are present. However, most components may appear to have nominally cyclic elastic stresses, but notches, welds, or other stress concentrations present in the component may result in local cyclic plastic deformation. Under these conditions, another approach that uses the local strains as the governing fatigue parameter was developed in the late 1950s. It has been shown to be more effective in predicting the fatigue life of a component, which is so called the strain-life approach.

The local strain-life method is based on the assumption that the life spent on crack nucleation and small crack growth of a notch component can be approximated by a smooth laboratory specimen under the same cyclic deformation at the crack initiation sit. Based on the proposal by Morrow (1968) the relation of the total strain amplitude (ε_a) and the fatigue life in reversals to failure ($2N_f$) can be expressed as in Eq. 9.

$$\varepsilon_a = \varepsilon_a^e + \varepsilon_a^p = \frac{\sigma_f'}{E}(2N_f)^b + \varepsilon_f'(2N_f)^c \tag{9}$$

where, σ'_f is the fatigue strength coefficient, ε'_f is the fatigue ductility coefficient, b is the fatigue strength exponent, c is the fatigue ductility exponent and $2N_f$ is the fatigue life in reversals. Equation 9 also known as the strain-life equation, is the basic expression for the strain-based approach for fatigue. This equation is the summation of two separate curves for elastic strain amplitude-life (ε^e_a-$2N_f$) and for plastic strain amplitude-life (ε^p_a-$2N_f$) Dividing the Basquin Eq. 2 by the modulus of elasticity gives the equation for the elastic strain amplitude-life curve:

$$\varepsilon_a^e = \frac{\Delta\varepsilon^e}{2} = \frac{\sigma_a}{E} = \frac{\sigma_f'}{E}(2N_f)^b \tag{10}$$

Both Manson (1953) and Coffin (1954) simultaneously proposed the equation for the plastic strain amplitude-life curve:

$$\varepsilon_a^p = \frac{\Delta\varepsilon^p}{2} = \varepsilon_f'(2N_f)^c \tag{11}$$

When plotted on log-log scales, both curves become straight lines as shown in Fig. 4. Besides the modulus of elasticity E, the baseline strain-life expression is defined by the four regression parameters: σ'_f, ε'_f, b and c.

The presence of nonzero mean stress can influence fatigue behavior of materials because a tensile or a compressive mean stress has been shown to be responsible for accelerating or decelerating crack initiation and crack growth. Experimental data have shown that compressive mean stresses are beneficial and tensile mean stresses are detrimental to fatigue life (Julie *et al*., 1990). Morrow (1968) has proposed the following relationship when a mean stress (σ_m) is present:

$$\varepsilon_a = \frac{\sigma_f' - \sigma_m}{E}(2N_f)^b + \varepsilon_f'(2N_f)^c \tag{12}$$

Equation 12 indicates the mean stress can be taken into account by modifying the elastic part of the strain-life by the mean stress (σ_m), as shown in Fig. 5.

Referring to Fig. 5, the model indicates that a tensile mean stress would reduce the fatigue strength coefficient (σ'_f) whereas a compressive mean stress would increase the fatigue strength coefficient. Equation 12 has been extensively cited for steels and used with considerable

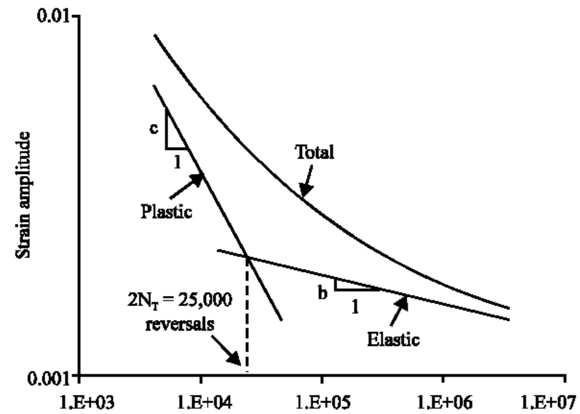

Fig. 4: Diagrammatic plot of a total strain-life curve

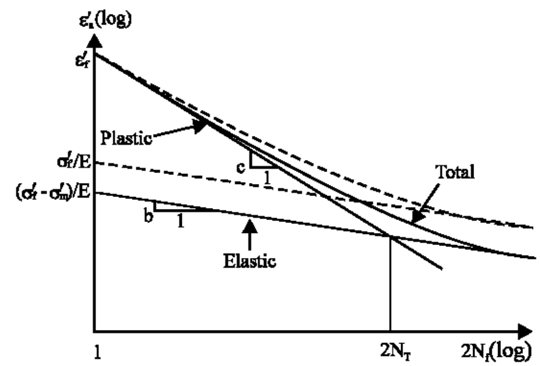

Fig. 5: Morrow's mean stress correction model

success in the long life regime when plastic strain amplitude is of little significance. In addition to the Morrow's strain-life model, Smith *et al*. (1970) proposed a method that assumes the amount of fatigue damage in a cycle is determined by $\sigma_{max}\varepsilon_a$, where σ_{max} is the maximum tensile stress and ε_a is the strain amplitude. The SWT parameter $\sigma_a\,\varepsilon_a$ for fully reversed test is equal to $\sigma_{max}\,\varepsilon_a$ for a mean stress test. Thus this concept can be generalized and expressed in the following mathematical form:

$$\sigma_{max}\,\varepsilon_a = \sigma_{a,rev}\,\varepsilon_{a,rev} \qquad \sigma_{max} > 0 \tag{13}$$

where, $\sigma_{a,rev}$ and $\varepsilon_{a,rev}$ are the fully reversed stress and strain amplitudes, respectively, that produce an equivalent fatigue damage due to the SWT parameter. The value of $\varepsilon_{a,rev}$ obtained from the strain-life curve of Eq. 9 and the value of $\sigma_{a,rev}$ from the cyclic stress-stain curve of Eq. 13. The SWT parameter predicts no fatigue damage if the maximum tensile stress becomes zero and negative. When materials behave ideally and satisfy the compatibility condition (i.e., $n' = b/c$ and $K' = \sigma'_f/(\varepsilon'_f)^{n'}$, the maximum tensile stress for fully reversed loading is given by:

$$\sigma_{max} = \sigma_a = \sigma'_f (2N_f)^b \quad (14)$$

When Eq. 14 is multiplied by the strain-life equation, then the SWT mean stress correction formula is expressed as following mathematical expression:

$$\sigma_{max}\, \varepsilon_a E = (\sigma'_f)^2 (2N_f)^{2b} + \sigma'_f\, \varepsilon'_f\, E(2N_f)^{b+c} \quad \sigma_{max} > 0 \quad (15)$$

This Equation is based on the assumption that for different combinations of strain amplitude ε_a and mean stress σ_m, the product, $\sigma_{max}\, \varepsilon_a$ remains constant for a given life. If σ_{max} is zero, Eq. 15, Predicts infinite life, which implies that tension must be present for fatigue fractures to occur. The previous equations have been used to handle mean stress effects. The SWT formula has been successfully applied to gray cast iron (Fash and Socie, 1982), hardened carbon steels (Wehner and Fatemi, 1991) and microalloyed steels (Forsetti and Blasarin, 1980). Equation 15 has been shown to correlate mean stress data better for a wider range of materials and is therefore regarded as more promising for metallic materials.

RESULTS AND DISCUSSION

It is well known that fatigue life predictions, based on the Wöhler curve from the constant amplitude tests, often give large systematic prediction errors for service loads at variable amplitude. For a fatigue life of 10 variable amplitude cycles of a steering rod, the maximum allowable stresses are 50 to 100% higher than the constant amplitude fatigue limit, depending on the spectrum shape (Berger *et al.*, 2002) and in the constant amplitude tests predict a life twice as long as predicted by variable amplitude tests (Johannesson *et al.*, 2005). Many refinements of the fatigue damage calculations have been suggested, especially mean stress corrections and also crack closure models. Another solution is to modify the load at an optimal R-value test with a specific service load (Gassner line), or calibrate the predictions with a few variable amplitude (relative Miner rule). Our work lies in the second category and the method is to study at different variable amplitude loads that are representative for the service, then they are the bases for the predictions. The main contribution of this paper is the method for prediction of the S-N and ε-N curves from different variable amplitude load spectra as well as to study the factors affecting that prediction.

The finite element technique was used for modelling and simulation of the case studied of the shell as a cylinder in three-dimension mesh showed in Fig. 6. For this purpose, the finite element analysis technique was being used for the modelling and simulation based on MSC Nastran/Patran analysis codes. The type of element used here is tetrahedron element (10 nodes). To study the fatigue life prediction in this analysis a shell with T/D < 0.1 (where T is the thickness and D is the diameter) and L/D > 10 (where L is the length) was used.

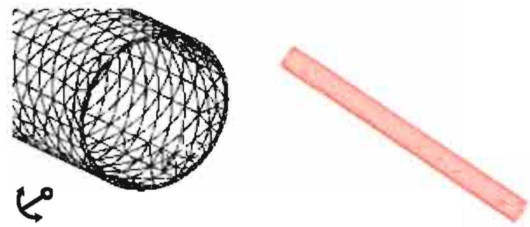

Fig. 6: Finite element mesh of a shell structure. Inner diameter = 250 mm, Thickness/diameter <0.1, Length > 10 diameter

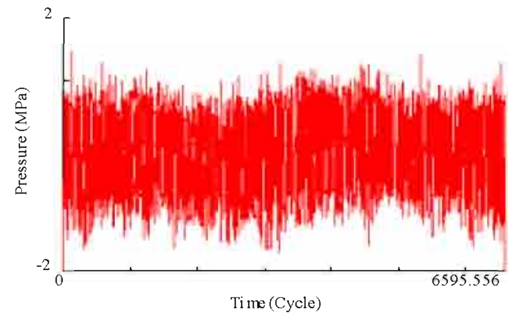

Fig. 7: SAE standard bracket loading history

Components or structures are subjected to quite diverse load histories, their histories may be rather simple and repetitive, at the other extreme, they may be completely random. Many of shell structures are subjected to complex cyclic loading spectrums ranging from small vibration to large load excursions. One of the three typical main load histories obtained by the SAE Fatigue Design and Evaluation, is the vibration load history, that was measured is dominated by essentially fully reversed loading, as shown in Fig. 7 and the typical CAL is shown in Fig. 8.

In this analysis, the material of the shell ASTM A533 steel has been used, for which the chemical compositions, mechanical properties at the room temperature are shown in Table 1 and cyclic properties are shown in Table 2. Figure 9 showed the strain-life curves with elastic, plastic and fatigue limit of this type of material.

In order to predict the life of a component subjected to a variable amplitude load history, it is necessary to reduce the complex history into a number of events. The counting method must count a cycle with the range

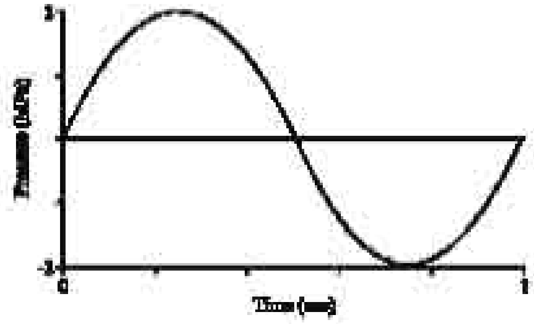

Fig. 8: Display of constant amplitude loading

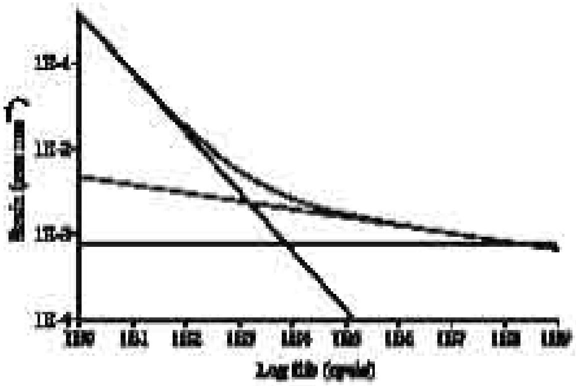

Fig. 9: The total strain-life curve for ASTM A533

Table 1: Chemical composition (wt%) and mechanical properties of A 533

C	Si	Mn	P	S	Ni	Cr	Cu
0.17	0.23	1.39	0.003	0.013	0.59	0.004	0.003

Co	Mo	V	Al	N	O	Fe
0.007	0.44	0.003	0.024	36 ppm	10 ppm	Bal.

Yield strength (MPa)	Ultimate strength (MPa)	Young's Modulus (GPa)	Elongation (%)	Reduction in area (%)
441	588	203	23.3	58

Table 2: Cyclic properties of the material (ASTM A533)

Fatigue strength coefficient (MPa) SF	975.300
Fatigue strength exponent b	-0.095
Fatigue ductility exponent c	-0.690
Fatigue ductility coefficient EF	0.350
Cyclic strain hardening exponent n'	0.110
Cyclic strength coefficient (MPa) K'	940.200

from the highest peak to the lowest valley and must try to count other cycles in a manner that maximizes the ranges that are counted. The most popular and probably the best method of cycle counting is rainflow method which is used here. Using the cyclic load history mentioned above as an internal pressure to the shell, to understand the evaluation and distribution of cyclic stress/strain field under cyclic loading conditions. The linear static analysis

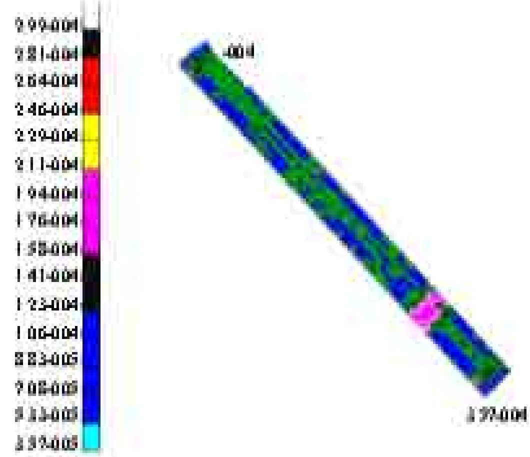

Fig. 10: Contour display of strains on shell structure

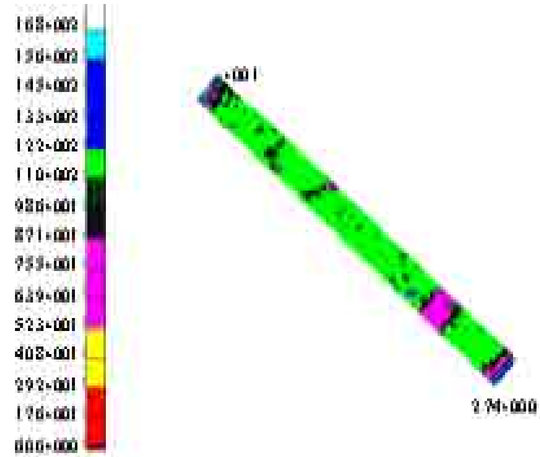

Fig. 11: Contour display of stresses on shell

was performed using finite element software to determine the stress and strain results for the model. The results of the maximum principle stresses and strains are used for the subsequent fatigue life analysis. Figure 10 and 11 showed the contour (image) of the stress and strain distribution on the shell.

In the study by Savaidis *et al.* (2002), cyclic loading components with constant amplitude have been assumed to simplify the analysis. Approximate model has been developed for a notched cylinder under fatigue loading with variable amplitude in many studies as a constant load to simplify the analysis. Figure 12 showed the log life of pressure vessel under VAL as well as the effect of using CAL on fatigue life prediction of the pressure vessel. From Fig. 12 the life predicted with CAL is larger than (10%) with VAL, but the slop of the curves seems to be same, which means that using CAL in steed of VAL to simplify the analysis cannot simulate the actual case. The

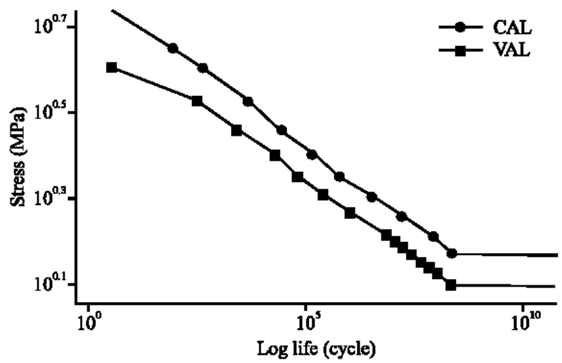

Fig. 12: Life prediction of shell with VAL and VAL

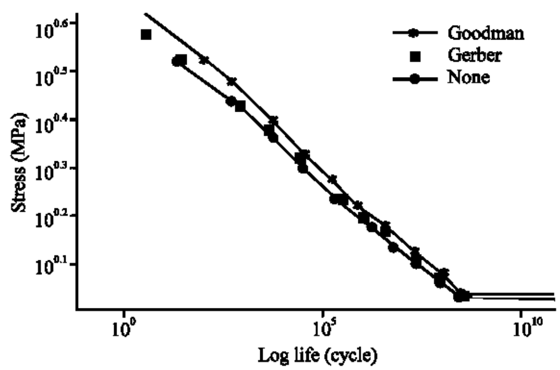

Fig. 13: Life prediction of shell with VAL and mean stress

normal mean stresses have a significant effect on fatigue behavior of components. Normal mean stresses are responsible for the opening and closing state of micro-cracks. The effect of mean stress on alternating fatigue life; i.e.,: no mean, Goodman and Gerber are studied and showed in Fig. 13.

Fatigue failure of mechanical components is a process of cyclic stress/strain evolutions and redistributions in the critical stressed volume. It may be imagined that due to stress concentration (notches, material defects or surface roughness) the local material yields firstly to redistribute the loading to the surrounding material, then follows with cyclic plastic deformation and finally crack initiates and the resistance is lost. The earliest fatigue life prediction approach used the stress range as fatigue parameter for life correlations, afterwards, the strain range was recognized as fatigue parameter for life prediction. The inclusion of mean stress effect in fatigue life prediction methods involving strain-life data is more complex. Several models are dealing with mean stress effect on strain-life fatigue behavior. The effect of using Morrow or SWT correlations in studying crack initiation, showed in Fig. 14, which gave small differences in the log

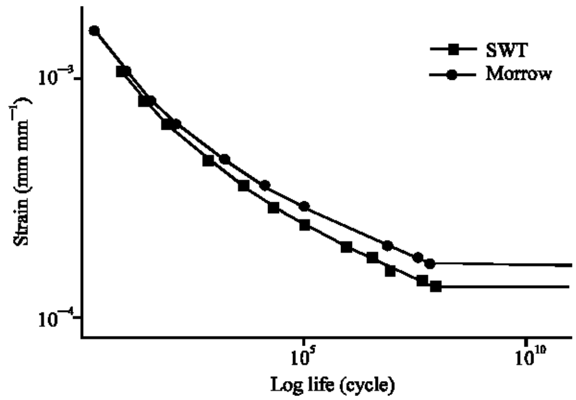

Fig. 14: Log life curve of shell

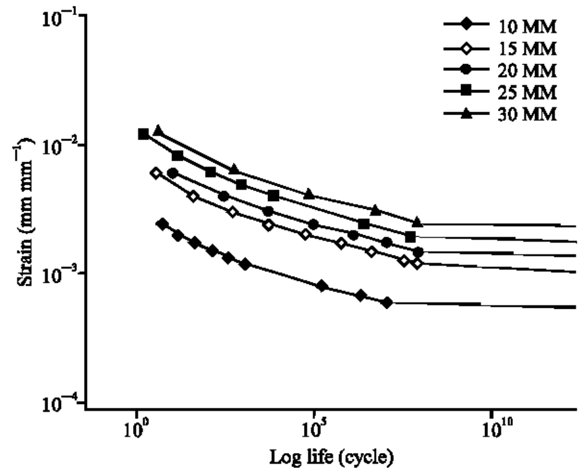

Fig. 15: Effect of thickness on life prediction of shell

life curve, although the SWT formula is as more promising for use. The two formulas take in to account the effect of mean stress.

To show the effect of the thickness on fatigue life of the shell. In this case study the inner diameter of the shell is 250 mm and the thickness was taken ranging from 10 to 30 mm as a critical thickness and it's effect can be shown clearly on the life of the shell. Figure 15 showed the increment of load capacity of the shell to 50% when the thickness changed from 10 to 15 mm with the same loading, while in the other steps are less and further increasing the thickness may goes with analysis out of shell theory.

Since most of fatigue failure originate at the surface, the surface will have a substantial influence on fatigue behavior. Surface effects are caused by differences in surface roughness, microstructure, chemical composition and residual stresses. Most engineering parts, however, are not highly polished and grinding or machining, even

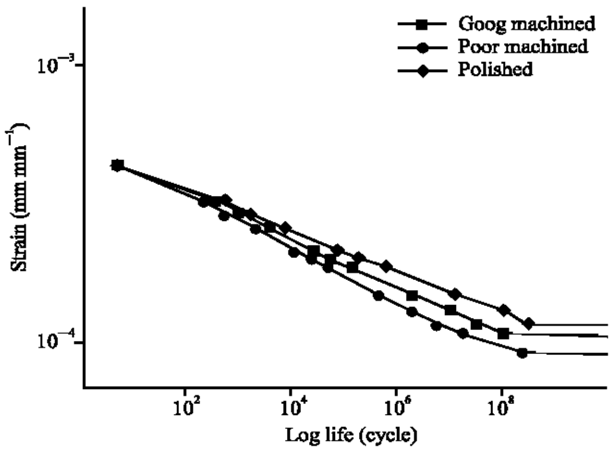

Fig. 16: Effect of surface roughness on the fatigue life of the shell

if done carefully, will cause degradation in fatigue strength. To show the effect of various surface finish on fatigue life of the shell, three types of surface finishing have been appeared in Fig. 16. The simulation showed the effect of surface finish on fatigue life of the shell as a comparison between the polished, good machined and poor machined, the polished surface gave better life at the same strain value with others.

CONCLUSIONS

The fatigue life prediction methods (S-N and ε-N) with different theories, the simulating and modeling of the shell using finite element analysis for variable amplitude loading was proposed in this study. A comparison of using CAL and VAL and the effect of mean stress, thickness of the shell and surface finish was discussed and a reasonable difference was found between them. The conclusion of this case study is that constant amplitude predict a life larger as long as one predicted by variable amplitude tests. However, no difference in the slope is seen. The difference is a multiplicative factor. It is our belief that the proposed method of using variable amplitude loading will lead to more accurate predictions, compared to constant amplitude.

This study showed how the mean stresses affects the fatigue life condition, which they are varied from curve (Gerber) to straight line (Goodman) as well as the no-mean stress. It was seen that the Gerber approach are tend to fall between the Goodman approach and no-mean curves, for most fatigue design situation R<1 (i.e., R values approaching 1), there is little differences in the theories.

The FEA used to study the crack initiation in shell cylinder under VAL with Morrow and SWT strain life equations showed with small divergence between them. Morrow accounted for the effect of the mean stress by modifying the strain-life curve, while the SWT strain-life model, the damage parameter is taken to be the product of the maximum stress and the strain amplitude of cycle. For loading sequences that are predominantly tensile the SWT approach is more conservative and therefore recommended. In a case of the loading being predominantly compressive, particularly for wholly compressive cycles, the Morrow model provides more realistic life predicted.

The crack initiation is easily affected by a number of material surface conditions and shell thickness. These effects on strain-life were shown which indicate that good surface finish and high shell thickness gives longer life.

When observing to the stages for improving the findings of this research, more calculation need to be done in order to optimize thickness of the shell with respect to loadings and materials. Also, there are many factors have influences on life prediction of pressures vessels need to take into account in future studies for example; corrosion, residual stresses, temperature change, material properties and heat treatment.

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